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Simulation on the New Thermal Storage in high temperature solar groove type thermal power

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Abstract: This paper presents an analysis of high temperature solar groove type thermal power utilizing high grade thermal energy. At night or in the rain day, the energy to power flash distillation is provided by a high temperature thermal storage device. The 3-D model of phase change heat transfer in thermal storage was founded by the software FLUENT, numerical simulation on heat storage and release performance of stainless steel plate-fin thermal storage filled with high temperature phase-change materials is performed. Analysis of the influences on the charge and discharge performance is carried out, which is caused by perforated-fin and serrated-fin, respectively. The inlet temperature and velocity of the heat-transfer fluid affected by solidification and melting rate of phase change material was analyzed. The thermal storing process of the plate-fin latent thermal storages is studied numerically. They also can provide good references for the design and optimization of new solar thermal storage

Keywords； simulation, solar groove type thermal power

1 Introduction

Under the background of increasingly stern situation in environmental pollution and energy crisis, solar energy has become the clean energy of the most attractive, the most greatly researched and developed and the most widely applied in all of new energy and renewable energy resources.[1-4] However, as the influences of low effective solar radiation arrived in earth, geographic location, such regular changes as diurnal variation and seasonal fluctuation, and some random factors, like weather changes etc, therefore, we should store the excess solar energy though related technologies and release it when it is scant, in order to meet the continuous and stable need in production and daily life.[5-7]As the most mature technology in the thermal storage, the technology of thermal storage with phase change has become a new focus in solar energy's utilization because of such advantages as high density in heat storage, the constant temperature in storing and releasing heat， easy control and so on.[8]

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In this paper,a high temperature solar phase-change thermal storage with fin-plate of phase-change was studied. The heat storage and release performance of thermal storage affected by the inlet temperature, flow rate, the fin-plate structure and the natural convection of heat fluid was emphatically studied using numerical simulation and experimental method. The main researching contents are as follows.

2. Plate-fin thermal storage model

This is a plate-fin thermal storage device (see Fig.1). The whole thermal storage device is made of stainless steel in order to ensure their high thermal conductivity. A phase change material $NaNO_3$ fills the stacked passages with fins above the clapboard. As a result, heat is supplied to the PCM along all three coordinate axes. Water is used as HTF for the numerical simulation and flows through the stacked passages with fins below the clapboard. Thus, it is assumed that heat is transferred among the PCM, $NaNO_3$ layer, the clapboard, the water layer and the HTF.

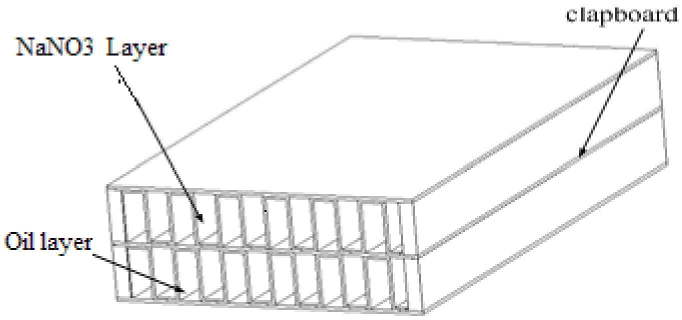

Fig.1 plate-fin thermal storage

Table 1 The main physical parameters of the PCM

Materials	$NaNO_3$
Density (kg/m^3)	2257
Specific heat (J/kg K)	2100
Thermal conductivity (W/m K)	1.5
Melt heat (J/kg)	174000
Melt temperature (K)	578

A physical model of the system is presented in detail as below. Then the computational procedure is discussed.A schematic view of the three-dimensional physical model is shown in Fig.1. The properties of aluminum and paraffin wax are summarized in Table 1

Consider three-dimensional periodic stacked passages with fins shown schematically in Fig.2. A theoretical model for phase changing in the plate-fin thermal storage is developed under the following assumptions:

(1) The PCM is homogeneous and isotropic. Solid specific heat, liquid specific heat, thermal conductivity, density and other parameters are constant, and does not change with the temperature.

(2) The natural convection caused by gravity is negligibly small.

(3) Thermal resistance across the wall of the thermal storage is negligibly.

(4) Heat loss from the thermal storage to its surroundings is negligibly.

(5)The phase-change process is considered to be unsteady-state and three-dimensional.

(6) The heat-carrying fluid (oil) is incompressible, and the channel within the flow is turbulent state.

(7) The convective heat transfer coefficient between the accumulator and the HTF is constant.

Due to the periodicity of the fins and clapboard in the thermal storage geometry, only a portion of the geometry will be modeled in Fluent, and the representative domain can be considered as in Fig.3.

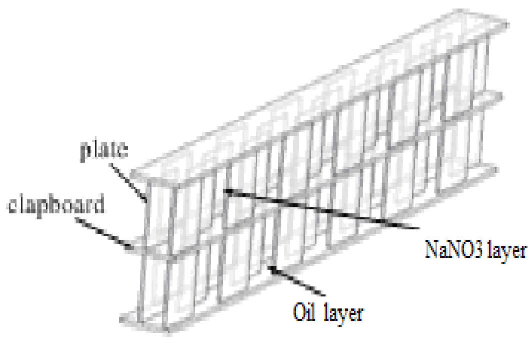

Fig 2 The dimensions and boundary of the Serrated-fin model

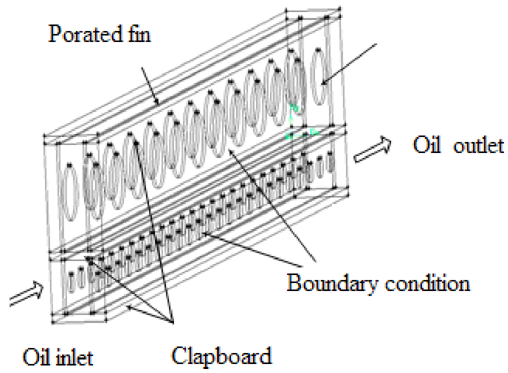

Fig.3 The dimensions and boundary of the Perforated-fin model

2.1 Governing equations

HTF region

Continuity:

$$\frac{\partial \rho_f}{\partial t}+\frac{\partial(\rho_f u)}{\partial x}+\frac{\partial(\rho_f v)}{\partial y}+\frac{\partial(\rho_f w)}{\partial z}=0 \tag{1}$$

Momentum：

$$\rho_l \frac{\partial k}{\partial t}+\rho_l u_j \frac{\partial k}{\partial x_j}=\frac{\partial}{\partial x_j}\left[(\mu+\frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial x_j}\right]+\mu_t \frac{\partial u_i}{\partial x_j}(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i})-\rho_l \varepsilon \tag{2}$$

$$\rho_l \frac{\partial \varepsilon}{\partial t}+\rho_l u_k \frac{\partial \varepsilon}{\partial x_k}=\frac{\partial}{\partial x_k}\left[(\mu+\frac{\mu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x_k}\right]+\frac{c_1 \varepsilon}{k}\mu_t \frac{\partial u_i}{\partial x_j}(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i})-c_2 \rho_l \frac{\varepsilon^2}{k} \tag{3}$$

Energy:

$$\frac{\partial(\rho_f T_f)}{\partial t}+\frac{\partial(\rho_f u T_f)}{\partial x}+\frac{\partial(\rho_f v T_f)}{\partial y}+\frac{\partial(\rho_f w T_f)}{\partial z}=\frac{\partial}{\partial x}\left(\frac{k}{c_f}\frac{\partial T_f}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{k}{c_f}\frac{\partial T_f}{\partial y}\right)+\frac{\partial}{\partial z}\left(\frac{k}{c_f}\frac{\partial T_F}{\partial Z}\right)+S_r \tag{4}$$

PCM region

In the PCM region, the basic equation is:

$$\frac{\partial(\rho_p H)}{\partial t}+\nabla(\rho_p v H)=\nabla(\lambda \nabla T)+S \tag{5}$$

Because the momentum source term and convection are both negligible, the equation is:

$$\rho \frac{\partial H}{\partial t}=\lambda \nabla^2 T=\frac{\partial}{\partial t}\left(\lambda \frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial t}\left(\lambda \frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial t}\left(\lambda \frac{\partial T}{\partial z}\right) \tag{6}$$

$$H=h+\Delta h \tag{7}$$

The *h* is defined as the sum of sensible enthalpy, $h = h_{ref} + \int_{T_{ref}}^{T} c_p dT$, where h_{ref} is the reference enthalpy at the reference temperature T_{ref} . The enthalpy change due to the phase-change βL , $\Delta h = \beta L$.

2.2 Meshing

Mesh is generated in software GAMBIT 2.3. In this study the hybrid grid is used, as shown in Fig.4 .

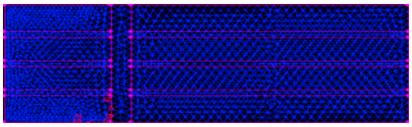

Fig.4 Serrated fin model grid

The simulations are carried out in a portion of plate-fin thermal storage. A three-dimensional computational domain is employed and divided into 555,520 cubic cells.

2.3 Boundary conditions

The surface of the regenerative heat exchanger is adiabatic:

$$\frac{\partial T}{\partial x} = \frac{\partial T}{\partial y} = \frac{\partial T}{\partial z} = 0 \tag{8}$$

Surface between hot media and the regenerator is for coupling boundary ,Coupling boundary has three representation：

1)Temperature continuous ： $T_w\Big|_f = T_w\Big|_p$ ， Among them, the f is for thermal fluid area, p representative regenerator area;

2)Heat flux continuous： $q_w\Big|_f = q_w\Big|_p$ ；

3)The third kind of coupling boundary conditions： $-\lambda \frac{\partial T}{\partial x}\Big|_{x=0} = k\left(T_w - T_f\right)$

2.4 Initial conditions

Initial temperature of the regenerative heat exchanger for thermal fluid and regenerator should be equal, because no heat is input or output, two area will automatically be balanced.

$$T_f\Big|_{(x,y,z)|t=0} = T_p\Big|_{(x,y,z)|t=0} = T_0 \tag{9}$$

3. Results and discussion

Storage rate or release rate of accumulator is only standard to determine the performance of the regenerator. In order to study heat storage and release characteristics of the new accumulator, the structure form of rib and entrance parameter of heat flow are researched in this paper. The main research contents are included as follow: the fluence of serrated fins and porous fins on heat storage / release; the fluence of entrance temperature of HTF on PCM melting rate of regenerator; the fluence of entrance temperature of HTF on regenerator PCM solidification rate; the fluence of HTF entrance velocity on the heat accumulator in PCM melting rate;the influence of HTF entrance temperature on the regenerator PCM solidification rate; compare different fin type on heat storage / release characteristics of different effects.

3.1Entrance temperature, speed of heat fluid on storage rate of the storage

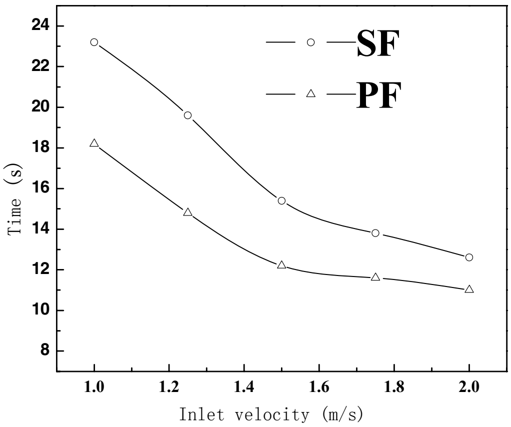

Fig5 PCM melting time with HTF entrance velocity change curve

For the above two kinds of structure in any one of a, in the regenerative process, along with the HTF entrance temperature increase in PCM complete melting time is shorter, and the temperature is increased from 585K to 595K time shortened obviously PCM complete melting time with HTF entrance speed changes, as can be seen from the graph, in the same HTF entrance temperature conditions, with the HTF entrance velocity increases, the PCM completely melted time becomes shorter and shorter, and with the entrance velocity increased melting time decreased slow; can be predicted when the accumulator structure size is given, when the entrance velocity to a certain value, the accumulator storage time will tend to be an approximate limit.

3.2 Entrance temperature, speed of heat fluid on coagulation rate of the storage

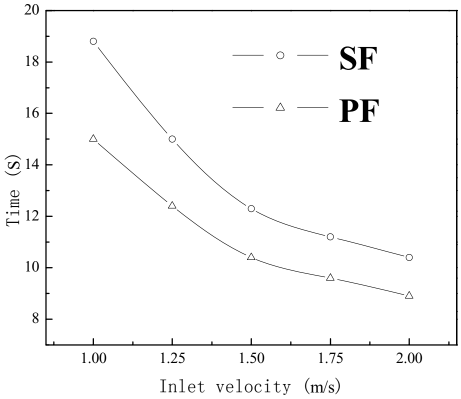

Figure 6 PCM coagulation time with HTF entrance velocity change curve

The two rib structure types on regenerator heat transfer effects were compared, the results found in the same entrance parameters, using a porous fin structure of regenerator heat time is the zigzag rib structure to be short; and in the same entrance velocity, in the calculation of temperature range within the two time difference with the temperature difference of heat transfer increases gradually, in accordance with the relevant heat transfer theory with the HTF entrance temperature increase heat melting time decreased gradually and tends to a very small value; in the heat transfer temperature difference under the same conditions, the two time gap as the entrance velocity increased gradually.

4.Conclusion

1. For solar phase-change heat storage device, using a porous fin structure of heat accumulator storage / release performance was better than that using serrated fin structure of heat accumulator. In the phase change heat transfer process in porous fin, heat transfer enhancement effect was significantly stronger in zigzag rib reinforcement effect.
2. With the HTF entrance velocity increases, and the PCM body temperature difference increases, the accumulator to complete the heat storage and heat release time is short.
3. Analysis results show that, in the HTF HTF entrance entrance velocity, temperature and PCM body temperature difference under the same conditions, the accumulator radiation time was significantly shorter than the storage time.

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